Comparisons between various cavity and panel noise reduction control in double-panel structures

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Abstract: This paper presents comparisons between various panel and cavity resonance control methods to reduce the transmitted sound in a double-panel structure. The double-panel, which consists of two panels with air in the gap, has the advantages of low weight and effective transmission-loss at high frequency. Therefore, it is widely applied in many areas such as aerospace. Nevertheless, the resonance of the cavity and the poor transmission-loss at low frequency limit its noise control performance. Applying active control forces on the panels or utilizing loudspeakers in the cavity to reduce the noise problem have been discussed in many papers. In this paper, an acoustic-structure coupled model is used to investigate and to compare the transmitted sound reduction of various cavity and panel resonance control methods. The control performance comparison is based on the same stability control margins. Moreover, an adaptive control method is used in the system to further improve the control performance. Piezoelectric actuators on the radiating panel in the adaptive feedforward control combines with the loudspeakers with pressure source in the feedback control is found to be the most effective combination.
Keywords: Double-panel, Panel control, Cavity control

1. Introduction

The increasing need for a comfortable environment points to the importance of noise control technology. With the development of smart materials and computation power, noise control is no longer only using passive control but also involving in many active control methods in the last decades. Passive control mainly means adding high damping materials or installing resonators to the system [1, 2]. However, due to the wavelengths property, passive control usually achieves less noise reduction and comes with a heavy implementation [3]. On the other hand, active control can provide potential advantages of reduced weight and better performance at lower frequencies. Active noise control (ANC) has been developed for decades and has found successfully applying in small spaces with broadband noise [4, 5]. However, for a large control region, this 3D computation problem will become very complicated and inefficient. Instead of dealing with 3D wave propagating problem, active structure acoustic control (ASAC) directly control the vibrating structure to reduce its radiating sound. This method can make the computation problem from 3D to 2D [4, 6]. The control strategy and the algorithm also have been investigated and designed for various applications. For a large configuration, decentralized control can effectively reduce the computation amount of the controller [7-11]. Decentralized feedback control strategy has been noted for its remarkable performance for the broadband objective[12]. A combination of direct feedback control and adaptive feedforward control can improve the performance of the broadband active noise and vibration control [13]. The adding weight of the controller installation is another

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important issue [14]. The double panel with an air gap structure can provide the advantage of a low weight structure, is another common noise reduction method [1, 15].

In this paper, the comparison between various combinations of sensors-actuators, direct feedback control, and adaptive feedforward control are analyzed. A real time structural control has been done to prove the numerical conclusion of our previous paper; piezoelectric actuators can only effectively reduce the transmitted noise when they are attached to the dominant resonant panel [16]. This paper is composed of three sections. First, the multiple decentralized feedback and adaptive feedforward control theory are introduced. Second, the finite element method model and the experiment measurement methods are described. Finally, the control performances of various control strategy combinations are compared and discussed.

2. Multiple decentralized control

2.1. *Feedback control loop*

A feedback control signal flow is shown in Fig. 1. $\mathbf{G}(j\omega)$ is the plant transfer matrix, $\mathbf{u}(j\omega)$ is the control signal matrix, and $\mathbf{d}(j\omega)$ is the noise source matrix. $\mathbf{H}(j\omega)$ is the control matrix, which is a constant H in this paper. $\mathbf{e}(j\omega) = (\mathbf{I}+\mathbf{G}(j\omega)\mathbf{H}(j\omega))^{-1}\cdot\mathbf{d}(j\omega)$, is the error signal matrix.

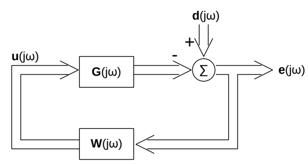


Figure 1. Direct velocity feedback system.

To present the interactions between multiple control units, the plant transfer matrix is fully coupled as,

$$\mathbf{G}(j\omega) = \begin{bmatrix} \mathbf{G}_{11}(j\omega) & \cdots & \mathbf{G}_{1m}(j\omega) \\ \vdots & \ddots & \vdots \\ \mathbf{G}_{l1}(j\omega) & \cdots & \mathbf{G}_{lm}(j\omega) \end{bmatrix} \tag{1}$$

where $\mathbf{G}_{lm}(j\omega)$ is the transfer matrix from the m^{th} actuator to the l^{th} sensor.

2.2. *Control stability*

In theory, the control stabilities can be unconditionally stable when the sensors and the actuators are collocated; otherwise the control gain is limited. By the Nyquist criterion, the MIMO decentralized control system is said to be stable when the plot of $\det[\mathbf{I}+\mathbf{G}(j\omega)\mathbf{H}(j\omega)]$ neither crosses nor encircle the origin (0, 0) [12]. However, in the presence of perturbations, the stable system can become unstable. The perturbation endurance of the system can be represented by stability margins. Gain margin, phase margin and modulus margin are used in this paper.

2.3. *Adaptive feedforward control*

Fig. 2 shows a feedforward control signal flow, where the noise source $\mathbf{d}(j\omega)$ is used as the reference signal.

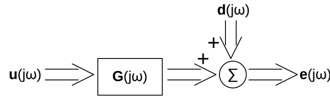


Figure 2. Feedforward system.

In Eq. (2), the cost function J is defined as the squared error signals plus the squared control signals with en effort weighting factor β. $\mathbf{e}^{\mathrm{H}}$ is the Hermitian matrix of $\mathbf{e}$. Steepest descent algorithm was used for the adaptive controller. The control signal can be derived as Eq. (3). Furthermore, the stability of the decentralized MIMO feedforward control is guaranteed while the real parts of the eigenvalues $\boldsymbol{\lambda}$ of the matrix $\hat{\mathbf{G}}^{\mathrm{H}}\mathbf{G}+\beta\mathbf{I}$ are positive. If the system is unstable, β can be set to $-\min \mathrm{Re}(\boldsymbol{\lambda})$ to make the system just stable [17]. However, feedforward control needs perfect knowledge of reference signal, therefore the control results may only apply for tonal noise.

$$J = \mathbf{e}^{\mathrm{H}}\mathbf{e} + \beta \mathbf{u}^{\mathrm{H}}\mathbf{u} \tag{2}$$

$$\mathbf{u} = -[\hat{\mathbf{G}}^{\mathrm{H}}\mathbf{G} + \beta\mathbf{I}]^{-1}\hat{\mathbf{G}}^{\mathrm{H}}\mathbf{d} \tag{3}$$

3. Model analysis and measurement

3.1. Acoustic-structural interaction FEM model

An acoustic-structural finite element model was built to analyze the characteristics of our system. The model had been proved that it can well present the interactions between the acoustic pressure in the fluid domain and the structural deformation in the solid domain [16]. Fig. 3 and Fig. 4 show the configurations of our simulation model and experiment configuration. The primary noise source was produced by an incident spherical pressure wave from the corner of the bottom side, which can produce an asymmetric incident noise wave. The double panel structure was modeled by two simply supported panels and a cavity with 35mm thickness. And hard-wall boundaries were set for these cavity side walls. Table 1 is the parameter list of this FEM model.

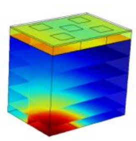


Figure 3. Acoustic-structure interaction model.

Figure 4. Experiment configuration.

Table 1 –Model parameters

	Parameters	Values	Unit
Aluminum panel	Dimensions	420*297*2	[mm^3]
	Density	2700	[kg/m^3]
	Young’s modulus	70	[GPa]
	Poisson’s ratio	0.33	
	Loss factor	0.03	
Honeycomb panel	Dimensions	420*297*5.8	[mm^3]
	Density	409	[kg/m^3]
	Young’s modulus	3.7	[GPa]
	Poisson’s ratio	0.33	
	Loss factor	0.03	
PZT patches	Dimensions	7.24*7.24*0.264	[mm^3]
	Density	7800	[kg/m^3]
	Young’s modulus	52	[GPa]
	Poisson’s ratio	0.33	
	Strain coefficient d_{31}	-190	[meter/Volt]
Acrylic box	Inner Dimensions	420*297*350	[mm^3]
	Wall thickness	40	[mm]
Middle cavity	Inner Dimensions	420*297*35	[mm^3]

3.2. Measurement and real time control

A double panel mounted on a rectangular box was set up for measurement. A loudspeaker in the bottom of the rectangular box generated the primary noise source. This box was made with 40 mm thick walls of acrylic plates to prevent the sound from leaking through side walls. The inner dimensions of the box were 420*297*350 mm^3. The primary noise source first entered an aluminum panel (the incident panel), then a layer of air of 35mm thickness followed by a honeycomb panel (the radiating panel). On each panel, there were both 5 velocity sensors and 5 piezoelectric patches on it. The kinetic energy of the radiating panel was measured by these 5 velocity sensors on the radiating panel.

4. Results discussion

4.1. Structural control with piezoelectric patches

Our previous paper showed that piezoelectric actuators can only effectively reduce the transmitted noise when they are attached to the dominant resonant panel. In the double-panel structure, the resonance comes from the resonant modes of the incident panel, the radiating panel, and the cavity. Therefore, both the incident panel and the radiating panel need to be controlled simultaneously [16]. Kinetic energy of the radiating panel can represent the radiating sound pressure level of the panel in the far field at low frequencies. Numerical analysis shows the kinetic energy frequency response of the radiating panel in our system (Fig. 5). In this frequency range, both incident panel and radiating panel dominate the resonant energy. In order to reduce all the resonant peaks, 10 piezoelectric actuators were used to control these two panels. However, the interaction of these two panels reduces the system stabilities. Nyquist plots of the control systems are shown in Table 2. The control gain is limited by the stability. The increasing complication of two independently controlled panels brings less stability; therefore the control performance of 10 piezoelectric actuators is also limited.

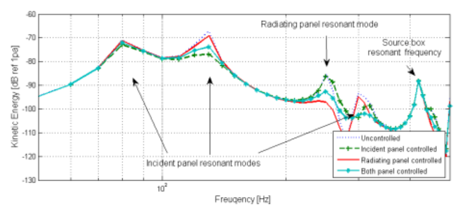
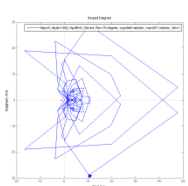
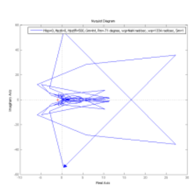
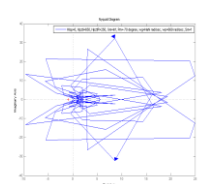


Figure 5. Piezoelectric actuators control performance.

Table 2 –Nyquist plot

incident panel	radiating panel	Both panels
		

In the real time control, 5 piezoelectric patches and 5 velocity sensors are attached to the radiating panel. Fig. 6 shows piezoelectric actuators can effective reduce all the resonant peaks except the source box resonant peak in the single panel structure. However, Fig. 7 shows radiating-panel piezoelectric actuators can only reduce certain peaks. The results of the real time control can prove the conclusions from the numerical analysis. Piezoelectric actuators can only effectively reduce the resonant peaks when they are attached to the dominant resonant panel.

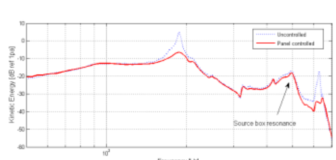


Figure 6. Single panel control result.

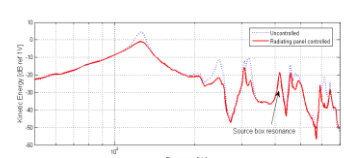


Figure 7. Double panel control result.

4.2. *Actuators and Sensors comparison*

Various sensor-actuator control strategies for direct feedback control in the double panel system had been analyzed. Three combinations are presented here. (1) 10 piezoelectric actuators and 10 velocity sensors (5 sets on each panel). (2) 6 loudspeakers with pressure source and 6 microphones. (3) 6 loudspeakers with acceleration source and 6 microphones. Fig. 8 is the configuration of the 6 loudspeakers and 6 microphones. Fig. 9 shows the control performance comparison, which is based on equal control gain margin, phase margin, and modulus margin (Table 3). The loudspeakers with pressure source feedback control can create more noise reduction in this double panel structure.

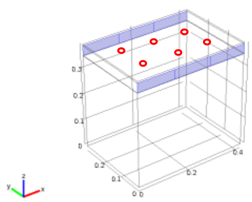


Figure 8. 6 Loudspeakers configuration.

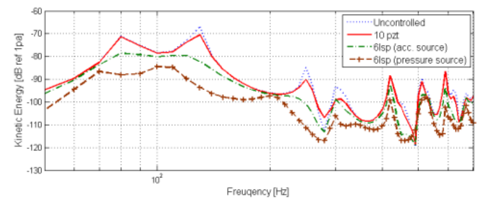


Figure 9. Control performance comparison.

Table 3 –Stabilities and energy reduction

Combinations	6 loudspeakers (acc. Source)	6 loudspeakers (pressure source)	10 pzt (inc. & rad. panels)
Control gain	0.001	0.77	265 (inc.); 150 (rad.)
Gain margin	Inf.	Inf.	Inf.
Phase margin	-76.2^{o}	-76.0 ^{o}	-76.1 ^{o}
Modulus margin	1.04	0.99	1.00
Total energy reduction* [dB]	**7.28**	**14**	**1.56**

*** (10*log$_{10}$(ΣKE_uncontrolled/ΣKE_controlled)**

4.3. *Feedback and feedforward combination*

Various actuators- sensors in feedforward control such as loudspeakers, piezoelectric patches on the radiating panel, and piezoelectric patches on both panels were chosen to be combined with various actuators-sensors in the feedback control such as loudspeakers, radiating-panel piezoelectric patches. With the noise reduction comparison between these combinations, we found piezoelectric patches on the radiating panel in the feedforward control combining pressure source loudspeakers in the feedback control can provide the largest improvement of the control effect. The combined control performance is shown in Fig. 10.

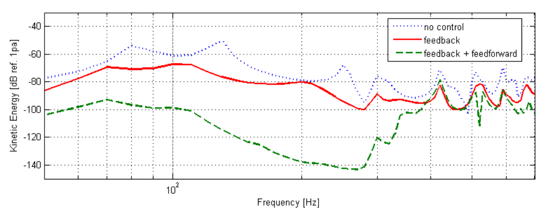


Figure 10. Combined control result.

5. Conclusions

Through the numerical analysis and the experiment measurement, this paper has shown that in the direct feedback control, piezoelectric actuators should be simultaneously applied to both the incident panel and the radiating panel in a double panel structure. However, the interactions between these two panels would reduce the control stability and limit the control performance. Loudspeakers with the pressure source can provide more noise reduction than panel attached piezoelectric patches in feedback control loop.

The combination of direct feedback control and adaptive feedforward control can further reduce the transmitted noise. From the comparison between various combinations, it shows piezoelectric actuators on the radiating panel in the adaptive feedforward control combines with the loudspeakers with pressure source in the feedback control can reach the lowest noise transmission.

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